Fermi National Accelerator Laboratory

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Operation and Maintenance of Fermilab's 300 kW Helium Screw Compressors after Two-Million Hours of Operation

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OPERATION AND MAINTENANCE OF FERMILAB'S 300 kW HELIUM SCREW COMPRESSORS AFTER TWO-MILLION HOURS OF OPERATION

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ABSTRACT

Fermi National Accelerator Laboratory uses 34 identical compressor systems connected to a common header to supply high pressure gas to the 24 satellite refrigerators of the Tevatron. These compressor packages consist of a two-stage oil injected Mycom helium screw compressor with associated oil removal system. This report summarizes (after 2,000,000 hours of operation) the problems, modifications and experiences associated with the operation of these compressor systems.

INTRODUCTION

The Tevatron refrigeration system uses thirty-four Mycom screw compressors installed in compressor buildings at nine locations around the Fermilab ring. Each of the thirty-four compressors consists of an oil-injected, two-stage, Mycom screw compressor driven by either a 260 or 300 kW (350 or 400 hp) motor and associated oil removal system. The system is designed to provide clean dry helium gas at 20 atm (~285 psig) to a 7.24 km (4-1/2 mile) header supplying 24 satellite helium refrigerators located around the Tevatron Accelerator.

Proper maintenance of such a large system is essential to reliable Tevatron operation. Typically, a minimum of 26 out of the 34 compressors are required during Tevatron operations. This redundancy in compressors allows maintenance to be carried on at all times regardless of the Tevatron status. This report will attempt to summarize after approximately 12 years and 2,000,000 total hours of operation, the problems, modifications and experiences associated with the operation of these compressor systems.

MAINTENANCE / EXPERIENCE

Compressor

The Mycom 2016C compressor consists of two sets of mating helically grooved rotors, a low stage and a high stage. The low stage male rotor is used as the driver and is coupled to the high stage rotor with a gear type coupling. Each male rotor consists of 4 lobes. The two female rotors consist of 6 lobes and are directly driven by the male rotors without the use of a timing gear. Both sets are made of heat resistant cast iron, a material that has proven difficult

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to repair. The rotors turn at 3600 revolutions per minute and are positioned with preloaded precision thrust angular ball bearings. Babbitt journal bearings support the rotors at each end. Capacity control is accomplished through the use of a sliding valve in the rotor casing which moves parallel to the rotor axis and changes the area of the opening in the bottom of the rotor casing. This, in effect, lengthens or shortens the compression zone of the rotor and further acts to bypass gas to the suction side. This mechanism is operated by a hydraulic piston and cylinder assembly located within the compressor itself. The piston is actuated by oil and driven in either direction according to the operation of a four-way solenoid valve. Operating experience has shown that if the high stage is loaded and the low stage unloaded, a vacuum develops between the stages. By adding a 20 percent minimum position metal spacer block on the low stage slider, positive pressure is retained with minimal increase in power. The slider hydraulic valves have been a high maintenance item over the years because of the cycling of the pistons. The constant cycling causes fatigue failure of stop washers within the valve assembly. This problem has been reduced by operationally locking the compressors in their fully loaded condition and only allowing a select number of compressors to regulate. Other areas which have shown significant wear are the slider guide groove and the guide blocks.

Main shaft seals are one of our lowest maintenance items when compared to industry, with an average of 21,000 hours per seal. Oil leakage is measured once a month. Total losses for all machines are less than 7.2 liters per month (7.625 quarts).

Over the last several years, Fermilab has been involved in a joint venture with Mycom Corporation in testing of an improved compressor at Fermilab[2]. Improvements to the compressor include a new modified rotor profile which reduces internal gas leakages, a new oil relief modification within the final compressed space reducing the oil compression within the compressor and a new generation lubricant with lower viscosity properties. These enhancements translated to a 58 kW reduction in shaft power while increasing efficiency by 33%. The results of these tests have paved the way for a complete retrofit of our entire compressor system to the new modified compressors. These modifications are tentatively scheduled to be incorporated within the next two years.

The main objective with any moving piece of equipment is to maximize the operating hours without damaging the device. In the first few years after commissioning, the compressors were regularly overhauled at the 20,000 hour mark as recommended by the manufacturer. Our operating statistics indicate 50,000 hours is a safe limit for the bearings.

Indications are that the second stage drive rotor thrust bearings determine the life of the machine. Our experience shows that the machine is in a self destruct mode when a bearing wears more than 0.0508 mm (0.002 in.). The normal supply of oil masks the vibrations, noise, and cutting action of the rotors striking the walls when the bearings fail. Higher power is the final message from an unattended, unmonitored machine. Inspection of several compressors with over 50,000 hours of service shows signs of spalling on the outer races of the second stage drive rotor thrust bearing. Because of this we have chosen 50,000 hours as the appropriate time to overhaul a compressor. A new or overhauled compressor typically has clearances of 0.0508 mm (0.002 in.) between rotors and 0.1778 mm (0.007 in.) rotor to housing. Inspections have shown that the high stage rotor to rotor backlash and rotor to housing clearance show significant wear after 50,000 hours corroborating the choice of 50,000 hours as a limit for these machines.

We use two methods to detect deteriorating bearings. A simple shaft contact probe placed on the second stage male rotor detects any axial movement of the shaft. When movement of the thrust bearing occurs contact is made between the second stage rotor and the shaft probe completing a circuit and subsequently tripping off the compressor. This device is set at 0.1016 mm (0.004 in.) to 0.1524 mm (0.006 in.). This works well but only protects the main drive rotor. Should this probe circuit fail and a bearing failure occur, the rotors would eventually strike the interior casing causing an increase in motor power and eventually tripping off the compressor and possibly damaging the motor. This device is a simple, valuable and inexpensive instrument preventing costly damage to the compressor.

The second method for detecting damaged bearings are monthly vibration readings taken with a hand held meter. Normal readings average 0.127 to 0.381 cm/sec rms. (0.05 to 0.15 in/sec rms.). Experience has shown that a compressor is in a self destruct mode when vibration readings consistently reach over 0.508 cm/sec rms. (0.20 in/sec rms.). Experience gained with the use of vibration readings has encouraged us to pursue the installation of accelerometers on one machine for testing.

To a limited degree we have fitted several compressors with thermocouples on each thrust bearing outer race, a total of four on each compressor. These temperatures are read continuously and registered on a chart recorder. During a thrust bearing failure at one of these locations, a rise of 1 to 2 degrees Celsius (~2 to 3 degrees Fahrenheit) was registered[3]. The temperature differential encountered was insufficient for incorporation of an alarm or shut down protection. Testing continues at the other installations.

Table 1 shows the overhaul history of all 34 compressors. Overhauls involve complete dismantling of the compressor and typically includes replacement of all wearing items such as bearings, gaskets, seals and rotors as needed depending on condition. Repairs done in place, such as shaft seal replacement which does not require complete dismantling of the compressor, are not considered overhauls and are not reflected in the overhaul statistics. The table indicates that of the 57 overhauls, roughly half can be attributed to bearing related problems and the other half to general 50,000 hour preventative maintenance removal. The remaining overhauls are attributed to gasket failure and damage caused by electrical failure. Compressors with partially blown high stage gaskets are typically kept in service. These compressors will run with a higher interstage pressure and run less efficiently but are typically not overhauled until the problem becomes severe enough to warrant removal. Surface grinding of the casing walls to make them flatter during overhauls has alleviated this problem. Several compressors have been damaged due to electrical failure of the motor. Although typically this involves compressor bearing damage it has been separated in the statistics to distinguish between the two.

Figure 1 shows the distribution of overhauls over the last ten years. In the first six years of operation all compressors were overhauled at 20,000 hours for preventative maintenance. This rate has slowed since the decision to overhaul compressors at 50,000 hours. The trend shows a decline in failures over the life of the machines. We believe that this can be attributed to experience gained, improved preventative maintenance and the reliability of our shaft sensor probes.

Drive Motor

The compressor drive is a 260 or 300 kW (350 or 400 hp) 3600 rpm horizontal induction motor with an open drip-proof enclosure. It is connected to the compressor by a double-flexing disc coupling of the spacer type. The motors are lubricated and vibration analyzed every six months. Whenever the coupling is removed, the motor shaft is checked for vertical ball bearing play. By placing a dial indicator on the shaft and lifting it, we can determine the condition of the bearing or bearings outer race housing. A new motor bearing has a 0.0508 to 0.1016 mm (0.002 to 0.004 in.) play. The bearing at the drive shaft end takes most of the wear. The manufactured tolerances of the bearing are a maximum of 0.1016 mm (0.004 in.) We replace any motor above that number if it has high vibrations. A rebuilt motor exceeding maximum tolerances is sent back to the vendor. Once a month vibration is measured with a handheld meter. Readings average 0.127 to 0.381 cm/sec rms. (0.05 to 0.15 in/sec rms.). Bearing case noise varies from machine to machine and seems to have no effect on the life of a motor bearing.

Table 1. Compressor Overhaul Summary

Reason for Overhaul	# of Overhauls
High Hours/Prev. Maint.	26
Bearing Related Problems	26
Blown High Stage Gasket	3
Damage caused by Electrical Failure	2
Total	57

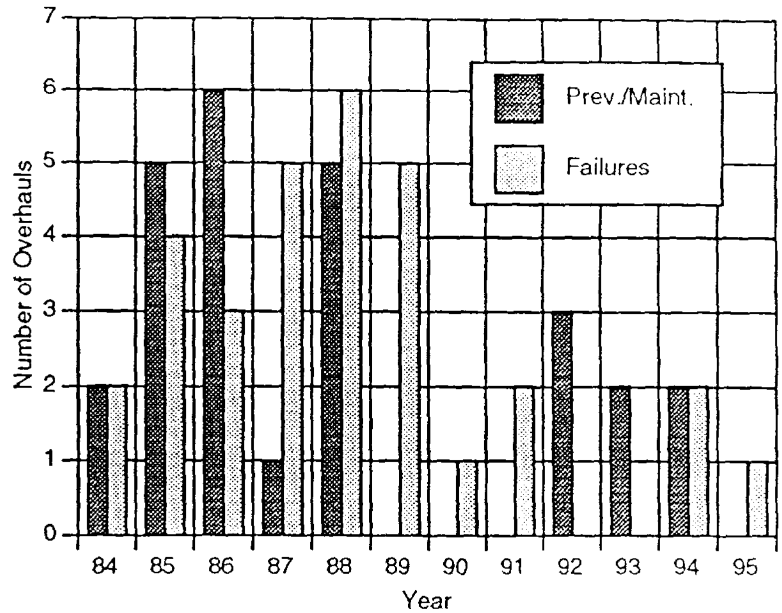

Figure 1. Compressor overhaul history

Since 1983, we have had 30 motors fail as figure 2 shows. Twenty-six have been attributed to winding failure with the remaining six failing due to bearing deterioration. Ten out of the thirty-four compressors are running with their original motors. Initially during the first few years of operation, many of the 260 kW (350 hp) motors were replaced with 300 kW (400 hp) high efficiency motors. Currently only seven of the 260 kW (350 hp) motors remain in service. Unlike compressors, the motors have been allowed to run continuously without periodic overhaul or reconditioning. The only maintenance performed are the monthly vibration readings and greasing every 6 months. Within the last few years we have begun to recondition motors which have over 80,000 hours. This limit was instituted after consulting with several bearing manufacturers, who define 100,000 hours as the limit to their bearings under normal operating conditions. Motor manufacturers are noncommittal in defining the life of a motor. As many of our motors were beginning to reach 70,000 hours of service we decided to begin reconditioning at 80,000 hours. Motor failures have been somewhat unpredictable as seen in figure 3. Figure 3 displays motor replacements with respect to location and indicates the number of hours each motor was in service with respect to the overall hours of the compressor skid. Certain locations have had as many as five motors fail while others are still running with their original motors. Some failures have occurred within as little as 1000 hours of initial operation while others have as many as 87,000 hours of operation without failure. This random failure rate may be attributed to differences in make and model of motors or perhaps in differences in starting conditions at time of failure such as high suction pressure causing the motor to run in its service factor, or several consecutive starts causing heat buildup and subsequent winding failure. In any case there seems to not be enough data to pinpoint weaknesses.

Lube Oil System

Oil plays an integral part in the performance of a screw compressor. It is required to seal, cool and lubricate the compressor as well as actuate the capacity control valves. The complete oil system consists of all the components necessary to store, circulate and cool the oil which is used as a lubricant and coolant for the compressor. Oil is stored in a pressurized vessel which serves as an oil reservoir and oil separator. This reservoir contains 340 liters (~90 gal.) of oil with an electric immersion heater keeping the oil above 15.6°C (60°F).

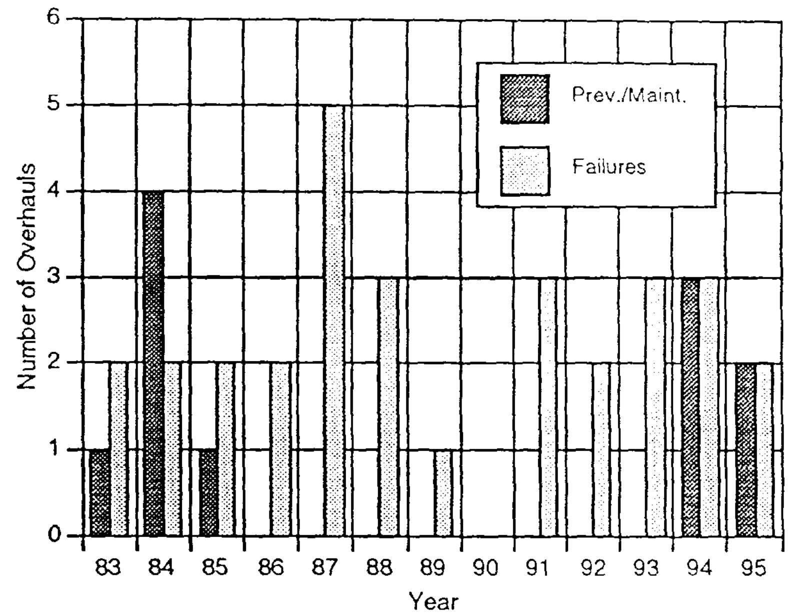

Figure 2. Motor overhaul history

An oil pump draws this oil through an oil cooler, a pump suction strainer, and drives it through a 10 micron filter and injects it into the compressor at several locations for bearing lubrication, heat dissipation and to provide a seal between the rotors and between rotors and the compressor housing. Oil is injected into the helium stream only during the first stage of compression and is discharged from the first stage entrained in the gas stream and then injected into the second stage. After the second stage of compression, the gas/oil mixture is delivered to the oil separator where the oil is allowed to settle out into the oil reservoir. Any oil remaining in the gas is further removed by an oil and moisture removal system downstream of the compressor.

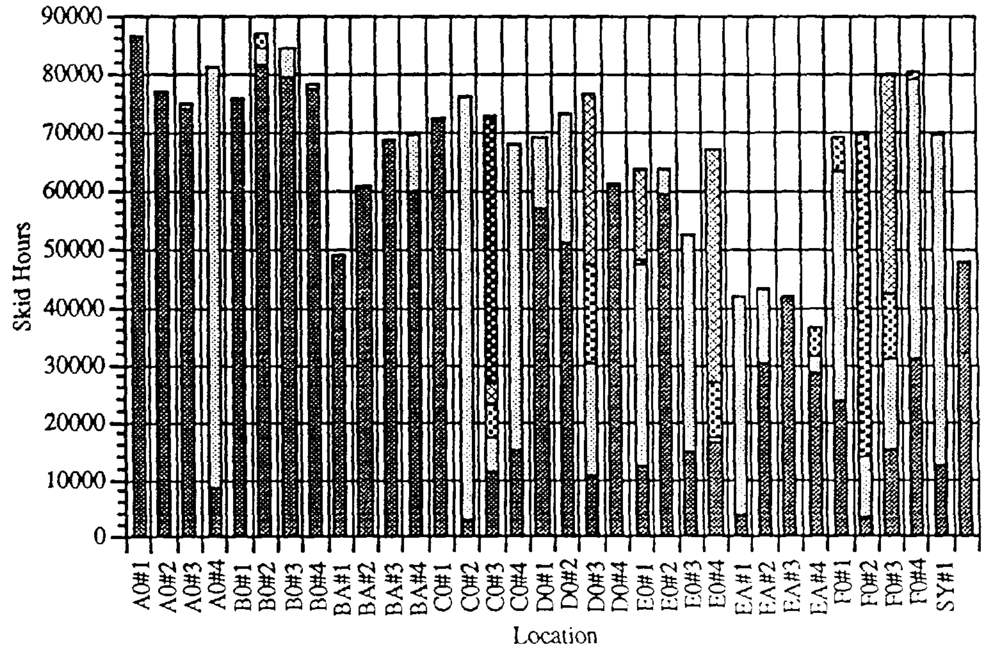

Figure 3. Motor replacement history

The oil inventory consists of Union Carbide LB-170X synthetic oil. With over 80,000 hours on many skids with original oil we have noticed no degradation in the oil except for some discoloration. Every two years selected compressors are sampled for viscosity and debris in the oil.

The vulnerability of the cryogenic system to contamination requires that the oil have a low water content in order not to cause plugging of heat exchangers and other equipment associated with the refrigeration system. The Union Carbide LB-170X oil as shipped from the manufacturer contains over 1500 parts per million of water by weight. Fermilab reprocesses the oil to reduce the water content to levels suitable for helium cryogenic service. Reprocessing consists of heating the oil to 121°C (250°F) and pulling a vacuum during the processing. When processing is complete the oil will contain less than one part per million of water by weight.

As mentioned earlier, Fermilab has been involved in a joint venture with Mycom Corporation in testing a new and improved compressor[2]. One of the improvements is in the use of a new generation lubricant with lower viscosity properties than the Union Carbide oil. Tests indicate that the use of this lower viscosity oil provides a reduction of approximately 14 kW in shaft power with a corresponding increase of 6.5% in isothermal efficiency and without any significant degradation in volumetric efficiency. The new oil (Gargoyle Arctic SHC 218HE) is a synthetic polyalphaolefin base manufactured by Mobil USA specifically developed for use in helium gas compressors.

Oil is circulated using an oil pump with a screw-configuration design and is rated at 300 LPM (80 GPM) of flow. The pumps have been a low maintenance item. Oil pump seals have averaged over 30,000 hours and require less maintenance than the main shaft seals. The oil pump is directly driven by a 5.6 kW (7.5 hp), 1750 rpm motor. These motors have an average of over 60,000 hours on them and are beginning to show their age. We are scheduling the replacement of these motors and plan on reconditioning all of the pumps within the next year.

Heat Exchangers / Water System

Oil cooling is achieved through an oil cooler heat exchanger where the cooling medium is water. The oil cooler is a four pass shell and tube heat exchanger with the water passing through the tubes and the oil through the shell. Water fouling of these heat exchangers is an on going problem. Every two years the heat exchangers are descaled using a biodegradable solution and the tubes are reamed as necessary.

The water system is a high maintenance area. Water erosion plays havoc with water valve seats and plungers. Recently the water valve plunger material has been changed from bronze to a plastic material with promising results. Water valve operation has been changed from gas ballast actuation to an electric actuated type. This modification allows more precise temperature control and also allows for periodic cycling of the valve to remove scale formation build up at the seat of the valve. Manual blowdown lines have also been added to assist in cleaning out strainers. In one location, at certain times of the year, the strainers on four compressors must be blown down at least once a week because of the influx of fresh water clams.

An aftercooler is installed at the discharge side of the compressor. It is used to cool the discharge gas before entering the oil removal system. Cooling of the helium is necessary to efficiently coalesce the oil out of the helium. Failure to properly coalesce the oil would result in oil carry-over to the refrigeration system. Such an event would be very time consuming to repair and unpredictable. The original aftercoolers were of a design which did not allow the water circuit to be cleaned out. Consequently, the water side of these heat exchangers had become blocked with clam shells and mud from the cooling water ponds. Most aftercoolers had lost their heat exchange properties. A new aftercooler has been purchased which reestablishes good heat transfer and allows for adequate scale removal. Installation of these new aftercoolers is currently underway.

Oil and Moisture Removal System

The purification system removes oil mist, oil vapor, water vapor, and particulates from the compressed helium. The units are designed with consideration of modularity, reliability and necessary redundancy. The system is designed for oil and moisture removal to below 0.1 part per million by weight and is discussed in detail by Satti[4].

The oil in the gas/oil mixture is initially allowed to settle out as it enters the oil separator/reservoir located downstream of the compressor discharge port. The reservoir contains a demister element which effectively removes the oil to 2500 ppm by weight. The next level of purification involves three stages of oil mist removal using Monsanto "Mist Eliminator" coalescers. The first two stages remove the oil mist to levels acceptable for cryogenic service while the third redundant stage is used as a guard coalescer. Solenoid valves installed at these units are used to recycle oil back to the compressor's interstage. A charcoal adsorbent vessel is installed downstream of the coalescers to remove any oil vapor entrained in the helium gas. The vessel contains 270 kg (~600 lbs.) of carbon pellets. A molecular sieve vessel is installed downstream of the charcoal bed and is used to adsorb water vapor left in the helium gas. Finally, a final filter is installed to retain any particle of 1 micron and larger which may have come from the adsorbent beds.

With over ten years of operation, the purification system has not shown any signs of oil carryover. Oil carryover at the third coalescer is monitored every six months

SUMMARY

The two-stage Mycom Corporation gas screw compressor has proven to be a reliable, low maintenance machine. The improvements initiated by Fermilab have contributed to this reliability. Upgrades such as the complete retrofit of existing compressors to new compressors with modified rotor profiles and new generation lubricant will greatly enhance the efficiency and performance of the compressor system. Experience gained over the years has drastically reduced the number of overhauls performed on the compressors. Much of this can also be attributed to the installation of the shaft movement sensor probe. The use of this probe has allowed technicians to take problem compressors off-line before catastrophic compressor failure can occur, thus saving time and money. Time invested in the design and testing of oil removal and gas purification resulted in the selection of components requiring virtually zero maintenance.

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